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(54) Title: AUTOMATIC VALVE CLEARANCE ADJUSTER

(57) Abstract: An automatic valve clearance adjuster comprising an internally screw-threaded housing (1); an externally screw-threaded screw member (3) extending within the housing; the screw thread (4) of the screw member (3) having an external form which is complementary to the thread form of the internal thread (2) of the housing and fits therein with a predetermined axial clearance, the thread being trapezoidal in form, symmetrical in axial cross-section and exhibiting equal frictional resistance against movement in both axial directions; wherein the flank angles, helix angle and number of starts of the screw thread are selected to ensure that the screw member will rotate and advance axially out of the housing solely under the influence of an axial force on its end within the housing; and the emerging end (7) of the screw member is adapted to work in conjunction with an adjacent component to receive therefrom a frictional resistance to rotation.

Title: Automatic Valve Clearance Adjuster

Description of Invention

This invention relates to an automatic valve clearance adjuster for a valve operating mechanism as may be used, for example, in an internal combustion engine; the adjuster comprising an internally screw threaded housing, an externally threaded screw extending within the housing, and force exerting means acting axially on the screw to cause it to rotate and advance outwardly of an open end of the housing to lengthen the adjuster; the co-operating thread forms of the screw and the housing being so configured that the screw will rotate freely and advance out of the housing under the bias of the said axially directed force.

When such a valve clearance adjuster is placed in a space between two relatively-movable components of a valve operating mechanism, with the housing engaging one component and the screw engaging the other component, the adjuster operates to take up clearances in the valve operating mechanism by extending, by advancement of the screw member out of the housing, until it fully occupies the space between the components. By way of example, the adjuster may be interposed directly or indirectly between a cam and a valve or a valve operating member. In another arrangement the adjuster may provide a fulcrum for a lever type of valve-operating member, the position of the fulcrum being varied by the adjuster to take up operating clearances.

The background to the present invention including certain examples of prior art, and aspects of the present invention itself, will now be described by way of example with reference to the accompanying drawings, of which:-

Figures 1, 2 and 3 illustrate possible dispositions of automatic valve clearance adjusters in valve operating mechanisms for internal combustion engines;

Figure 4 is a cross-sectional view, on an enlarged scale, of the screw-threads of an adjuster which may be in accordance with the invention;

Figure 5 shows an automatic valve clearance adjuster in accordance with the invention, a portion of cam follower and a diagrammatic representation of the relevant cam position, where the adjuster is required to take up clearances in the valve train;

Figure 6 shows the valve clearance adjuster of Figure 5, and an indication of the relevant cam position, where the adjuster has just eliminated clearances in the valve train;

Figure 7 shows the valve clearance adjuster of Figures 5 and 6, a portion of the cam follower and a diagrammatic indication of the relevant cam position, where the valve train is just beginning to open a valve;

Figure 8 shows an automatic valve clearance adjuster in accordance with the prior art, the adjuster having just taken up clearance which may previously have existed at one or more positions such as A, B, C, and D in the engine valve train;

Figure 9 shows the adjuster of Figure 8, in its condition when the valve is being opened;

Figure 10 shows an automatic valve clearance adjuster according to a further embodiment of the prior art;

Figure 11 shows the valve clearance adjuster of Figures 8 and 9, with the screw displaced laterally within the housing of the adjuster;

Figure 12 shows the adjuster of Figures 8 and 9, with the screw angularly displaced within the housing;

Figure 13 is a fragmentary view of contacting flanks of mating screw-threads having a relatively low flank angle;

Figure 14 is a fragmentary view of contacting flanks and mating screw-threads having a relatively high flank angle;

Figure 15 is a fragmentary view of mating screw-threads illustrating the relationship between axial and radial thread clearance;

Figure 16 illustrates a position an adjuster in accordance with the invention may assume prior to assembly into an engine;

Figure 17 shows an automatic valve clearance adjuster in accordance with a further aspect of the invention, ready for assembly into an engine.

In all the claims, corresponding reference numbers are used for corresponding parts.

Referring firstly to Figures 1, 2 and 3, in each of these figures there is depicted an automatic valve clearance adjuster comprising a housing 1, an internal screw thread 2 extending into the housing, and a screw member 3 having an external screw thread 4 co-operating with the internal thread 2 of the housing. Reference numeral 5 indicates an axial force applied by spring means to the screw member at the end 6 thereof which is in the housing: preferably such force is applied by a compression spring engaging the screw member as described hereafter. Reference numeral 7 indicates the end of the screw member which protrudes outwardly from the housing 1 and which engages a co-operating part of the valve operating mechanism in order to transmit a valve-operating force or provide a reaction force, depending on the nature of the valve operating mechanism, which is necessary when the valve is to be opened against the force of its closing spring by the operation of a cam on a camshaft.

In the example shown in Figure 1, the adjuster is self-contained and is stationary in a socket in the body of the engine. The protruding valve actuating

end 7 of the screw member 3 has a domed end with a spherical surface, said domed end fits inside a cavity 9 in a cam follower 8. The cavity 9 has an ogive or conical cross section and therefore makes a narrow circular band of contact with the domed end 7. Rotation of the cam 10 makes the cam follower 8 oscillate about the fulcrum provided by the domed end 7 and thus actuates the engine valve via the valve stem 11.

In Figure 1 the housing 1 is shown with an open lower end which would be suitable if the force 5 were provided by oil pressure. For the force 5 to be provided by a spring the housing would have a closed end, as described hereafter.

Referring to Figure 2, this shows an adjuster incorporated in a "bucket" type of cam-follower; the housing 1 is preferably integral with the bucket 13. The valve actuating end 7 of the screw member 3 is shaped to make with the valve stem 14 an area of circular, annular or conical contact. In this design of valve actuator mechanism the bucket 13 is slidably mounted in a bore in the body of the engine. The cam 10 operates directly via the bucket 13 and the adjuster to impart downward valve operating movement to the valve 14.

In the example shown in Figure 3 the adjuster is self contained and acts as a moveable push-rod between the cam 15 and a rocker-arm 17. The adjuster is slidably mounted in a bore in the body of the engine. The adjuster transmits motion from the cam 15 to the rocker-arm 17 which can pivot about the axis 41 and so impart downward valve operating movement to the valve 18. The adjuster is fitted with a pressure pad 16 between the rocker-arm 17 and the valve actuating end 7 of the screw member 3. The pressure pad can slide axially within the housing 1 but is restrained against rotation therein.

In addition to the designs described above, the adjuster could be incorporated in other types of valve train. For example the adjuster housing 1

could be an integral part of a rocker-arm in a rocker-arm/push-rod mechanism.

The known prior art will now be discussed by reference to Figures 8, 9, and 10, each of which shows a valve clearance adjuster which mainly comprises a screw within an internally threaded housing as previously mentioned. In each case the valve clearance adjuster is shown with a schematic representation of one type of engine valve train mechanism with which said adjuster is in working relationship. The reasoning to be put forward applies to any other valve train mechanism.

In one category of prior art there are several variations on a simple working principle in which the aforementioned screw is not acted upon by an axial force but is rotated and advanced out of the housing under the action of a torsion spring. This is explained by reference to Figure 10 which shows the cam 10 of an engine, said cam being shown in an angular position which would allow the engine valve to close as shown at 39. Figure 10 also shows that the torsion spring 29 has rotated the screw 30 so that the said screw has advanced out of the housing 38 to eliminate any clearance which may previously have existed at one or more positions such as A, B, C and D in the engine valve train. Moreover, Figure 10 shows that there is an unbroken path of compressive force transmission from the base 31 of the housing 38 through the contacting screw threads (as shown for example at 33) and then through positions A, B, C and D to the top of the stem 34 of the engine valve. Therefore, differential thermal expansions of the various engine parts can produce a situation in which the valve is prevented from closing, (as shown for example at 40 in Figure 9) thus leading to component wear and reduced engine performance. This is a serious disadvantage in all valve clearance adjusters which function under the action of a torsion spring.

In another category of prior art (GB-A-2033472, EP-T-0032284, GB-2160945, GB- 2211263, WO-A1-89/05898, W0-90/10786, W0-90/10787) the aforementioned internal and external threads are of buttress formation; this is a common feature throughout this category. Such a configuration is depicted in Figure 8 in which the screw 3 and the housing 1 are shown in axial cross section. The buttress thread flanks 24H and 24S which are shown with an inclination G_R to a line perpendicular to the axis are termed "running flanks". The buttress thread flanks 25H and 25S which are shown with an inclination G_L to a line perpendicular to the axis are termed "locking flanks". The helix angle is Z.

Figure 8 depicts the engine cam 10 in an angular position which would allow the engine valve 11 to close as shown at 39. It also shows that the compression spring 22 has pushed the running flanks 24S of the screw into contact with the running flanks 24H of the housing and has advanced the screw 3 out of the housing 1 to eliminate any clearance which may have existed previously at one or more of positions such as A, B, C, D in the engine valve train. The advancing of the screw 3 out of the housing is hereinafter referred to as "take-up" movement.

Figure 9 depicts a situation in which the engine cam 10 has turned into an angular position in which it exerts a force which is reacted by the stem 11 of the valve and by the screw 3 of the clearance adjuster. The reaction on the valve stem has opened the valve as shown at 40; the reaction on the screw has forced the locking flanks 25S of the screw through the clearance (26 Figure 8) and into contact with the locking flanks 25H of the housing. In theory, the assembly of screw 3 and housing 1 should then behave as a solid in resisting the said reaction because rotation of and consequent retraction of the screw into

the housing should be prevented by the enhanced frictional resistance which results from the relatively high angle of inclination G_L of the locking flanks.

Summarising, the use of buttress threads should, in theory, ensure that under the action of axially applied forces there is a high resistance to the screw 3 being pressed into the housing and a low resistance to the screw being advanced out of the housing;

Practical trials have shown that attempts to put the above described principle of operation into practice are attended by several difficulties, which are described as follows:

It is essential that there should be a certain predetermined axial clearance between the threads on the screw and the threads in the housing because, as shown at 26 in Figure 8, it ensures that there is no obstruction to valve closure. To ensure that the screw runs freely within the housing it is also necessary to have radial clearance between thread crests and adjacent thread roots, as shown at 19 in Figure 4. It is therefore possible for the screw 3 to be displaced laterally and so become eccentric to the housing 1 as shown in Figure 11; or to become tilted relative to the housing as shown in Figure 12. With the screw in either of these positions, when the running flanks of the screw threads and the running flanks of the housing threads are pushed towards one another under the action of the spring 22, there is little tendency for the screw to be returned to the concentric position if the generator 35 (Figure 11) of the surfaces of the running flanks is too near to perpendicularity to the line of action of the force exerted by the spring 22. An elementary illustration of this is given in Figures 13 and 14. Figures 13 and 14 are enlargements of the zone labelled F in Figures 11 and 12. In the example shown in Figure 13 the flank angle G_R is drawn with a small angle of inclination. Figure 14 shows another example and is drawn with G_R having a larger angle of inclination. The

reference numerals 36 and 37 indicate elements of the running flanks 24S of the screw and the running flanks 24H in the housing respectively. Under the same frictional conditions in each of the two examples, a force P could produce the movement Q more readily in the example shown in Figure 13 than in the example shown in Figure 14. The movement Q is necessary to re-centralise said screw within the housing 1.

Thus, as observed in practical trials, when the angle of inclination of the running flanks is low the screw 3 is likely to remain in an unfavourable position in the housing 1, as shown in Figure 11 or Figure 12. in these unfavourable positions, as shown in Figures 11 and 12, the running flank of the thread on the screw makes only line or point contact with the running flank of the thread in the housing, and contiguous "surface-to-surface" contact through an oil film (i.e. boundary lubrication conditions) cannot take place and consequently there is a high frictional resistance which opposes, and in most cases prevents, "take-up" movement. Although it is common practice to lubricate the threads with oil under pressure from the engine's lubrication system, under the aforementioned point or line contact conditions the required contiguous oil film lubrication cannot be maintained, friction therefore becomes excessive, and so "take-up" movement is prevented or at best becomes sluggish.

Conversely, when valve opening forces bring together the locking flanks 25S of the threads on the screw and the locking flanks 25H of the threads in the housing, see Figure 9, the angularity of the locking flanks is favourable for centralising the screw within the housing. Contiguous oil film lubrication conditions can then exist and a movement (termed "back-off" movement) of the screw 3 into the housing 1 can take place momentarily, that

is until the compressive force acting between the engaging locking flanks breaks down the oil film and allows metal-to-metal contact to take place.

Summarising, in spite of the theoretical high friction/low friction properties of the locking flanks/running flanks respectively, the "back-off" movement can be too high and/or the "take-up" movement can be too low. If, in each valve closing/valve opening cycle, the "back-off" movement exceeds the "take-up" movement, the screw 3 retracts progressively into the housing 1 and contact is lost between the members of the valve train mechanism.

As well as the shortcomings in performance, as just described, there are difficulties in manufacture and there are problems concerning service durability when buttress threads are used. These problems are described as follows:

The aforementioned axial clearance (26 & 9 in Figures 8 and 9 respectively) has to be held within close limits by close control in the manufacture of the internal and external threads. If said clearance is too high, valve operation will be unsatisfactory and noisy. Referring to Figures 8, 9, and 15, typical values of G_L and G_R are 75° and 15° respectively. Giving the radial clearance (KM in Figure 15) the symbol C, the axial clearance is therefore: -

$$JK + KL = (C \text{ x tangent } 15) + (C \text{ x tangent } 75)$$

i.e. C multiplied by 4. This means that, for example, a tolerance of 0.1mm on axial clearance would require the combined diametral tolerancing on the internal and external threads to be set at about .05mm. This is an exceedingly difficult requirement in mass production.

A similar situation exists in regard to wear of the thread surfaces during service. A small increase in radial clearance due to surface erosion or flattening of surface asperities on the sloping flanks will cause a fourfold addition to the axial clearance and will again lead to unsatisfactory and noisy valve operation.

The provision of grooves and ridges on the locking flanks of the screw was disclosed in GB-2211263. This modification in the design was introduced in an attempt to keep the "back off" movement within acceptable limits; the intention of the design was that the ridges, having a reduced surface area, would more easily break down the oil film between the engaging locking flanks and would therefore effect an earlier metal-to-metal condition, thus reducing the magnitude of the "back-off" movement. There are disadvantages with this design:-

a. The manufacture of locking flanks with grooves and ridges is difficult.

b. As a result of the decreased area of contact, the rate at which wear occurs on the engaging locking flanks is increased with a consequent increase in axial clearance as explained in the previous paragraph.

The close diametral tolerances on the threads of the components of an adjuster (i.e. on the screw and in the housing) necessitate even closer tolerances in the manufacture of the tools which are used in the production of said components. This applies particularly to the formation of the threads on the taps which produce the internal threads in the housings. Also, the proportions of the buttress cross-section require the teeth of the taps to be unusually wide, this can cause undesirably high tapping torques.

To try to increase the efficiency of engines which have automatic valve clearance adjusters, emphasis is being placed on reducing friction between the contacting surfaces of the cam 10 and cam-follower 8.

One object of the present invention is to provide an improved construction of mechanical valve clearance adjuster, with particular reference to:

a. improving the reliability of "take-up" movement;

b. reducing friction between cam and cam follower;

c. avoiding the unfavourable ratio of radial clearance / axial clearance which occurs between internal and external mating threads when the said threads have a buttress formation, and which therefore necessitates very close manufacturing tolerances and also causes a rapid increase in axial clearance between the mating threads.

The valve clearance adjuster, according to one aspect of the present invention, comprises an internally threaded housing and, within said housing, a screw member having an external thread with a form which is generally complementary to the internal thread form of the housing and fits therein with a predetermined axial clearance; the thread being trapezoidal in form, symmetrical in axial cross section and exhibiting equal frictional resistance against movement in either axial direction; the flank angles, helix angle and number of starts in the screw thread being determined to ensure that the screw member will rotate and advance axially out of the housing solely under the influence of an axial force on its non-emerging end; the emerging end of said screw member being configured to work in conjunction with an adjacent component of, for example, the valve train of an IC engine and to receive from said adjacent component a frictional resistance to rotation; the non-emerging end of said screw being configured so that it assists axial movement of the screw member when it is acted upon by the aforesaid axially directed force.

One possible reason for sluggish "take-up" movement is the tendency for the screw 3 to move into a position of non-concentricity with the housing 1, as shown in Figures 11 and 12. In the present invention this tendency is reduced by increasing the angle G_R of the slope of the running flanks as already discussed with reference to Figures 13 and 14.

In order to avoid the need for very close tolerances on the diameters of the screw threads, it is very desirable to effect a substantial reduction in the ratio:-

Axial clearance ÷ Radial clearance = tangent G_L + tangent G_R

as already discussed with reference to Figure 15. As an example an increase in G_R from 15° to 30° and a reduction in G_L down to a value of 30° from the value of G_L = 75° which has been taken as an example, the ratio axial clearance ÷ radial clearance is reduced from (tangent 15° + tangent 75°) to (2 x tangent 30°), i.e. from 4 to 1.15.

Thus, in the present invention, the flanks of each thread on the screw member and in the housing are preferably inclined at an angle of 30° on each side of a perpendicular to the axis of the screw thread when viewed in axial cross section. These flank angles are found in modern threads which are used in standard bolts and nuts etc., and are commonly termed 60° threads. This has the advantage that for inspection purposes commonly available devices can be used; for example, microscope graticules, thread measuring needles, thread measuring machines, gap gauges and optical projector screens.

Thus the angle G_R of the running flanks equals the angle G_L of the locking flanks, i.e. $G = G_R = G_L = 30°$. Giving the helix angle the symbol Z and the coefficient of friction the symbol μ, the condition for the screw to rotate and advance axially under axial load is given by the formula: -

tangent $Z > \mu \div$ cosine G

this assumes negligible friction on the end of the screw which end receives the axial force. As approximate examples, a screw with an outside diameter of 8mm and a lead of 4mm, would rotate and advance under axial load if the coefficient of friction had a value less than approximately 0.14. Increasing the lead to 5mm would enable satisfactory operation to be achieved with an even

higher coefficient of friction, viz. 0.17. The design of screw threads according to these examples could be achieved satisfactorily by the use of two-start threads.

Thus it is possible to have a screw member which has a thread of symmetrical form with flank angles G of 30° and which can rotate and advance axially solely under the influence of axial force. In addition, the 30° flank angle enables the screw member to remain concentric relative to the housing and so avoid the point contact conditions which occur when lower flank angles are used, as described earlier. "Take-up" movement can thereby be achieved under the influence of an axial force. Such an axial force may be provided, for example, by the pressure of oil from the engine's lubrication system or by means of a small spring preferably acting on the end of the screw through the medium of a ball-ended plunger.

Practical trials have shown that when a screw member is made with a symmetrical "V" form, said screw member can be given sufficient resistance to "back-off" movement by providing an external frictional torque resistance at the end of the screw which end contacts a co-acting member of the engine valve train.

According to the design of the engine valve train, the zone of contact between the end of the screw member and the co-acting member of the valve train can be in the form of, for example:-

a. a circular line

b. an annular area

c. an area of the curved surface of a thin conical frustum

d. an area of the curved surface of a thin slice of a sphere.

In the case of b. c. or d. the width dimension may be small enough to justify the assumption that the contact condition is a line of circular contact. In all cases the axis of the circle, perpendicular to its plane, coincides with the axis about which the screw member could rotate. In each of the four examples a value D could be assigned to represent the diameter of the circle to which frictional resistance acts tangentially. Giving the mean diameter of the screw thread the symbol d, external frictional resistance to "back-off" movement is largely dependant upon the ratio D ÷ d and can therefore be controlled by design of:-

I. mean diameter of the screw

II. the valve actuating end of the screw member

III. the part of the member of the valve train which part contacts the screw member.

By way of example, engine tests have shown that an adequate control of "back off" movement of an adjuster is obtained when a ratio of D ÷ d = 2 is used in conjunction with a screw thread having an helix angle of 10°. This in turn gives a reliable and responsive "take up" movement.

The mode of operation of the adjuster in accordance with the invention will now be described with reference to Figures 5, 5A; 6, 6A; and 7, 7A which by way of example correspond with Figure 1. Figure 5 shows the screw member 3 in a notional position within the housing 1 when clearance has developed between members of the engine valve train. For ease of description the entire amount of clearance is shown as a single gap 42 between the end 7 of the screw member 3 and the cavity 9 in the cam follower 8. In this situation the cam 10 can be in any angular position in which its constant radius portion is contacting the cam follower 8, this is indicated diagrammatically in Figure 5A.

Owing to the relatively high helix angle and to the low frictional torque resistance between the ball 29 and the spherical depression in the end 6 of the screw member, the spring 22 is able to produce "take-up" movement, i.e. to advance the screw member 3 (upward as illustrated) out of the housing 1 until the spherical end 7 contacts the surface of the cavity 9 in the cam follower 8, as shown in Figure 6.

With regard to the situation shown in Figure 6, three points should be noted:-

a) The cam 10 can still be regarded as being in any angular position in which any point on its constant radius portion is contacting the cam follower 8, as shown diagrammatically in Figure 6A.

b) A clearance gap 43 is maintained below (as illustrated) the threads of the screw member 3 as was the case in the situation shown in Figure 5.

c) The force exerted by the spherical end 7 on the surface of the cavity 9 in the cam follower 8 is small. Said force is limited to the force in the spring 22 minus the effort required to produce the "take-up" movement.

Figures 7 and 7A show the situation where the cam 10 has turned into a position where it has pushed the cam follower 8 just sufficiently to move the screw member 3 downwards through the previously existing clearance gap 43. That is to say the clearance gap is now above (as illustrated at 44) the threads of the screw member 3. At this instant between the mating threads there will be a continuous oil film and consequently there will be from within the housing 1 a low resistance to upward "back-off" movement of the screw member 3. Resistance to "back-off" movement is therefore provided externally by friction

at the contact between the end 7 of the screw member and the surface of the cavity 9 in the cam follower 8.

Subsequent slight rotation of the cam 10 induces a rapidly increasing load throughout the entire valve train in order to overcome the force with which the engine valve is held against its seating. The aforementioned oil film between the mating threads is broken down and friction at the threads contributes an internal resistance to "back-off" movement.

With a given thread geometry and frictional conditions the aforementioned external resistance to "back-off" movement is largely dependant upon the ratio D ÷ d,

where D = effective diameter of the circle of frictional contact between the end 7 of the screw and the surface of the cavity 9 in the cam follower 8.

d = mean diameter of the screw thread.

The required resistance to "back-off" movement is therefore obtained by

design of
I the screw thread (in terms of its effective diameter),
II the valve actuating end 7 of the screw member 3
III the part of the member of the valve train which part contacts the end 7 of the screw member 3.

Also, increased resistance to "back-off" movement can be obtained by increasing the angle W (Figure 6), i.e. the angle between a transverse plane and a common tangent to the contacting surfaces of the cavity 9 and the spherical end 7.

Further rotation of the cam 10 opens the engine valve, then allows it to close, and then brings the valve train back to a situation corresponding to Figure 5 or Figure 6.

When an automatic valve clearance adjuster in accordance with the invention is not incorporated in a valve operating mechanism, i.e. not assembled in an engine, and hence is not constrained by the components it engages in the mechanism, the force exerting means may cause the screw member to advance outwardly of the housing to its maximum extent. In this condition the adjuster cannot be assembled into the engine, and requires to be shortened by retraction of the screw member into the housing by rotational movement relative thereto, until the overall length of the adjuster is sufficiently reduced for it to be placed between the relevant components. For assembly of a plurality of such adjusters into their respective positions in the individual parts of the valve operating mechanism for an engine having a plurality of valves, clearly it is desirable that there should be some means for temporarily holding the adjusters in their contracted condition so that such assembly is facilitated.

It is broadly the object of another aspect of the present invention to meet the above described requirement for temporarily holding an adjuster in its contracted condition.

According to another aspect of the invention, therefore, I provide a valve clearance adjuster comprising a housing having an internal screw thread, a screw member extending into the housing from an open end thereof and having an external screw thread engaging the thread within the housing, and spring means acting on the screw member in the direction of its longitudinal axis; the co-operating screw threads of the screw member and housing being of such a configuration that the screw member will rotate and advance out of the housing under the influence of the spring means; wherein there is provided abutment means operable between the screw member and the housing when the screw member is screwed into the housing to an inner position, whereby frictional forces can be established between the screw member, housing and

abutment means sufficient to retain the screw member in said inner position against the action of the spring means.

In a valve clearance adjuster in accordance with this aspect of the invention, it is possible for the screw member to be screwed into the housing to its inner position at which the abutment means operable between the screw member and the housing is engaged. Then if the screw member is tightened to cause an increased force to be exerted between the screw member, abutment means and housing, frictional forces will be established therebetween of sufficient magnitude to resist the action of the spring means which tend to advance the screw member outwardly of the housing. The dimensions of the valve clearance adjuster should be arranged so that when the screw member is in its inner position the overall length of the adjuster is sufficiently small to enable it to be easily assembled in the required position in the operating mechanism for a valve of an engine: hence an engine with a plurality of the valve clearance adjusters can easily be assembled. Tests have shown that on starting the engine, the shock induced in each adjuster by impact from the cam which operates its particular valve is sufficient to free the frictional lock between the screw and housing, so that the screw member immediately takes up the correct working position relative to the housing of the adjuster.

The screw member of the adjuster may be adapted to be engaged by a tool by which it can be screwed into the housing to cause the abutment means to be bought into operation as aforesaid. In embodiments of adjuster described hereafter, wherein the screw member has a head with a part-spherical surface for engagement with a complementary part-spherical socket formation in a rocker arm or the like, a tool for engagement with the screw member may comprise a socket or recess with a surface configuration able to establish frictional engagement with the head of the screw member sufficient to enable it

to be tightened to engage the abutment means when the screw is in its inner position.

The abutment means may comprise a shoulder formation provided inside the housing and engageable by an innermost end face of the screw member when the latter is in its inner position. Such a shoulder may be afforded by the housing itself, or a separate member, e.g. a sleeve, affording such a shoulder may be inserted in the housing.

Alternatively, the screw member may be provided with an abutment formation which is engageable with an end face of the housing. Such a formation may comprise a collar provided on the screw member beneath a head part thereof. Such a collar may also be usable for holding the screw member to screw it into the housing and tightening it when the collar has engaged the housing, rendering it unnecessary for a separate tool to be used in this case.

Preferably the valve clearance adjuster is in accordance with the first aspect of the invention. However, it may alternatively be an adjuster incorporating a buttress thread such as referred to in the above discussion of the prior art.

Figure 16 shows an adjuster such as illustrated in Figure 5, 6 or 7, in the condition it can assume when not assembled into an engine. The screw member has advanced out of the housing, so the overall length of the adjuster will be substantially greater than the space available to be occupied by the adjuster in an engine.

For assembly of the adjuster into an engine, therefore, the screw member 3 has to be screwed into the housing 1 to the extent that the overall length of the adjuster is reduced to a magnitude permitting easy assembly. Such overall length of the adjuster will be slightly less than the typical operating lengths of the adjuster shown in Figures 6 and 7.

If the characteristics of the screw thread operative between the screw member and housing of the adjuster are such that the screw member will not stay in such a position by friction, in accordance with the invention there may be provided abutment means operable between the screw member and housing when the screw member is in an innermost position similar to that in which it is shown in Figure 5. One form of such abutment means is shown in Figure 17, afforded by a cylindrical sleeve 50 disposed within the housing and surrounding the spring 22. One end of the sleeve 50 engages the closed lower end 51 of the housing, while the other end of the sleeve abuts end face 6 of the screw member in the annular region surrounding the recess therein.

When the screw member is screwed into the housing and tightened into engagement with the abutment sleeve 50, forces are established between the screw member and sleeve and between the cooperating threads of the screw member and housing for the screw member frictionally to be held in such an innermost position, despite the force exerted thereon by the spring 22. This enables the adjuster easily to be installed. However, when the engine in which the adjuster is installed is started, impact from the cam is sufficient to free the frictional lock between the screw member and housing, so the screw immediately takes up the correct working position as above described.

Instead of the sleeve 50, the housing could be provided with an integral abutment formation with which the screw member is engageable when it is in its innermost position. Such a formation could be provided, for example, by having the interior of the housing of a smaller diameter in its part occupied by the spring 22, beneath the lower end of its internal screw thread. Yet a further possibility is that the screw member could be provided with an external collar beneath its head, engageable with outer end face 55 of the housing. Such a

collar may be used as a means for turning the screw when it is to be screwed into the housing until the collar abuts the latter.

If the screw member is not provided with such a collar or other formation enabling it to be screwed into the housing in this way, a tool such as that indicated at 52 in Figure 17 may be used to engage the part-spherical head end 7 of the screw member to turn it. Tool 52 has a tapering socket 53 at one end whose angle is such that when pressed into engagement with the head of the screw member by hand it will frictionally grip the latter with sufficient force to screw the screw member into the housing against the action of spring 22 and engage the abutment sleeve 50. Tool 52 has a grip portion 54 enabling it to be held and turned easily by a person using it.

As above referred to, in some designs of adjuster incorporating spring means for advancing the screw member axially out of the housing, there may not be a need for the expedient above described to facilitate assembly of the adjuster into an engine. As above referred to, engine tests have shown that adequate control of adjuster "back off" movement is obtained with a ratio of D/d of 2 is used, in conjunction with a screw thread having an helix angle of 10° which in turn gives a reliable and responsive "take up" movement of the adjuster. However in cases where overall engine design considerations impose constraints which necessitate lower values of D/d, a lower thread helix angle has to be considered. For example, an adjuster which has to have a helix angle of 6°, which should produce adequate "take up" movement when it is subject to effective lubrication in an engine, should have sufficient thread friction to keep the screw in an inner position within the housing without the need for the assembly device.

In the present specification "comprise" means "includes or consists of" and "comprising" means "including or consisting of".

The features disclosed in the foregoing description, or the following claims, or the accompanying drawings, expressed in their specific forms or in terms of a means for performing the disclosed function, or a method or process for attaining the disclosed result, as appropriate, may, separately, or in any combination of such features, be utilised for realising the invention in diverse forms thereof.

CLAIMS

1. An automatic valve clearance adjuster comprising an internally screw-threaded housing (1); an externally screw-threaded screw member (3) extending within the housing; the screw thread (4) of the screw member (3) having an external form which is complementary to the thread form of the internal thread (2) of the housing and fits therein with a predetermined axial clearance, the thread being trapezoidal in form, symmetrical in axial cross-section and exhibiting equal frictional resistance against movement in both axial directions; wherein the flank angles, helix angle and number of starts of the screw thread are selected to ensure that the screw member will rotate and advance axially out of the housing solely under the influence of an axial force on its end within the housing; and the emerging end (7) of the screw member is adapted to work in conjunction with an adjacent component to receive therefrom a frictional resistance to rotation.

2. An automatic valve clearance adjuster according to Claim 1 wherein the flanks of each thread on the screw member and in the housing are inclined at an angle of 30° to the perpendicular to the axis of the screw thread.

3. An adjuster according to Claim 1 or Claim 2 wherein the energising end of the screw member is adapted to contact said adjacent component in one of:-

(a) a circular line
(b) an annular area
(c) an area of the curved surface of a thin conical frustum, and
(d) an area of the curved surface of a thin slice of a sphere.

4. An adjuster according to Claim 3 wherein, in the case of contact with said adjacent component in b, c or d, the area of contact approximates to line contact.

5. An adjuster according to Claim 3 or 4 wherein the diameter of said contact line or area is greater than the mean diameter of the screw thread in the adjuster.

6. An adjuster according to any one of the preceding claims comprising spring means acting on the screw member to urge it to advance axially out of the housing.

7. An adjuster according to Claim 6 wherein said spring means acts on the end of the screw member within the housing through a ball-ended member.

8. An adjuster according to any of Claims 1 to 5 wherein said screw member is adapted to be acted on by oil pressure to urge it axially out of the housing.

9. An adjuster according to Claim 6 or Claim 7 wherein there is provided abutment means operable between the screw member and the housing when the screw member is screwed into the housing to an inner position, whereby frictional forces can be established between the screw member, housing and abutment means sufficient to retain the screw member in said inner position against the action of the spring means.

10. A valve clearance adjuster comprising a housing having an internal screw thread, a screw member extending into the housing from an open end thereof and having an external screw thread engaging the thread within the

housing, and spring means acting on the screw member in the direction of its longitudinal axis; the co-operating screw threads of the screw member and housing being of such a configuration that the screw member will rotate and advance out of the housing under the influence of the spring means; wherein there is provided abutment means operable between the screw member and the housing when the screw member is screwed into the housing to an inner position, whereby frictional forces can be established between the screw member, housing and abutment means sufficient to retain the screw member in said inner position against the action of the spring means.

11. An adjuster according to Claim 9 or Claim 10 wherein said abutment means comprises a shoulder formation inside the housing and engageable by an innermost end face of the screw member.

12. An adjuster according to Claim 11 wherein said shoulder is afforded by the housing.

13. An adjuster according to Claim 11 wherein said shoulder is afforded by a member disposed within the housing.

14. An adjuster according to Claim 9 or Claim 10 wherein the screw member has an abutment formation engageable with an end face of the housing.

15. An adjuster according to Claim 14 wherein said formation comprises a collar provided on the screw member beneath a head part thereof.

16. An adjuster according to Claim 15 wherein said collar provides for holding the screw member for screwing it into the housing.

17. An adjuster according to any one of Claims 9 to 16 wherein the screw member is adapted to be engaged by a tool for screwing it into the housing and causing said abutment means to be brought into operation.

18. The combination of an adjuster according to Claim 17 and said tool engageable therewith.

19. The combination according to Claim 18 wherein said tool is frictionally engageable with said head portion of the screw member.

20. A valve operating mechanism including an automatic valve clearance adjuster according to any one of Claims 1 to 17 interposed between respective components of the mechanism for taking up clearances therebetween.

21. An internal combustion engine including a number of valve operating mechanisms according to Claim 20.

22. An automatic valve clearance adjuster substantially as hereinbefore described with reference to Figures 5 to 7 or Figure 17 of the accompanying drawings.

23. Any novel feature or novel combination of features described herein and/or in the accompanying drawings.

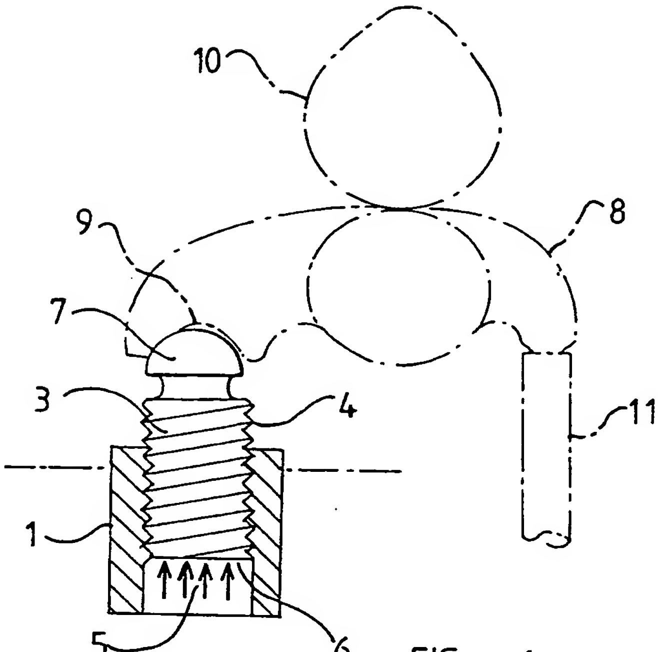


FIG 1

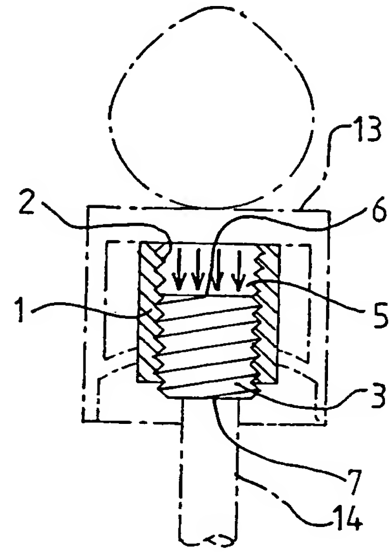


FIG 2

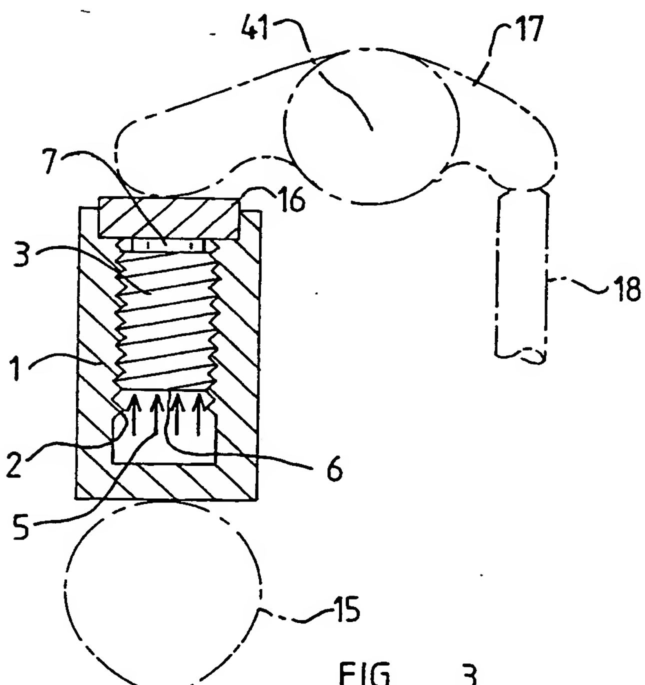


FIG 3

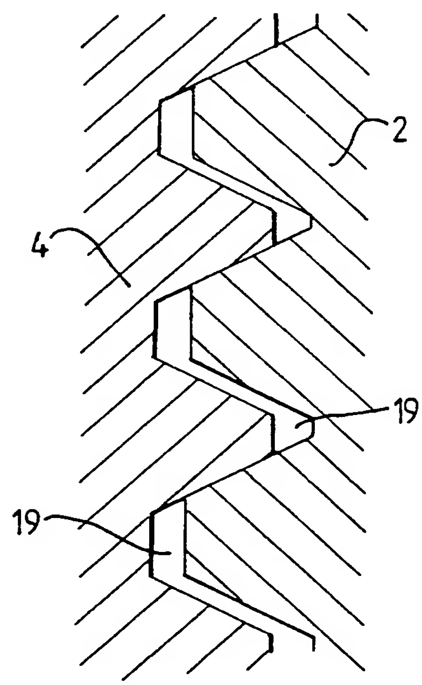


FIG 4

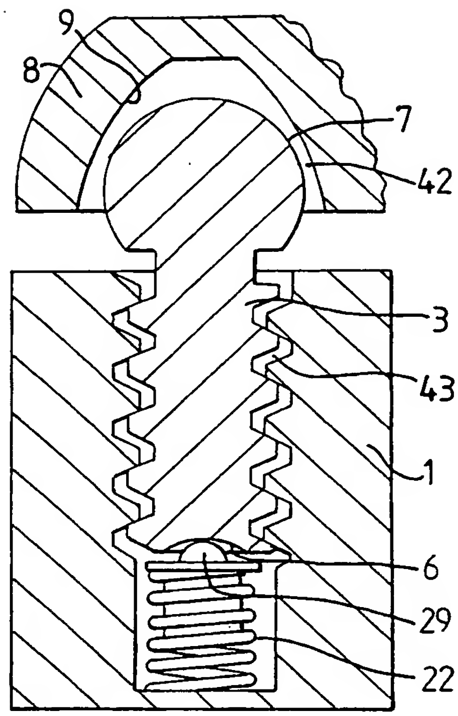


FIG 5

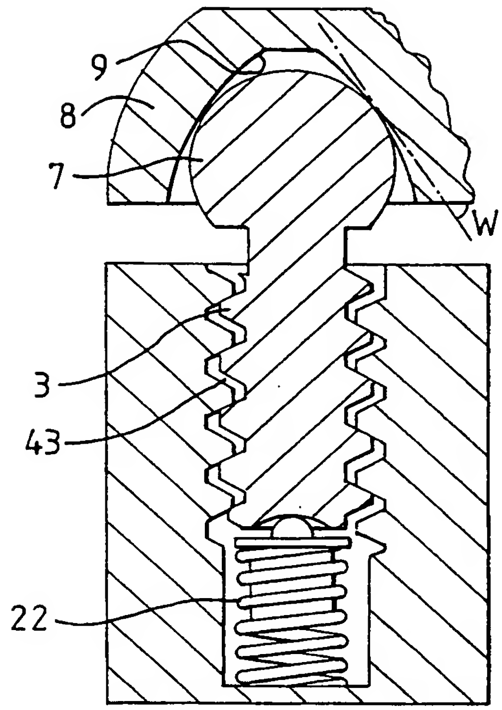


FIG 6

FIG 5A

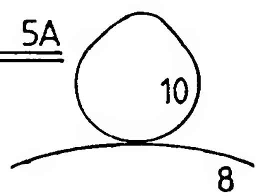


FIG 6A

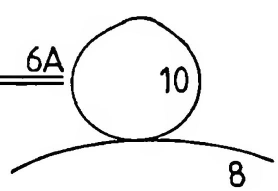
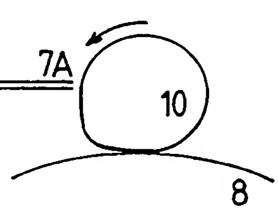


FIG 7A



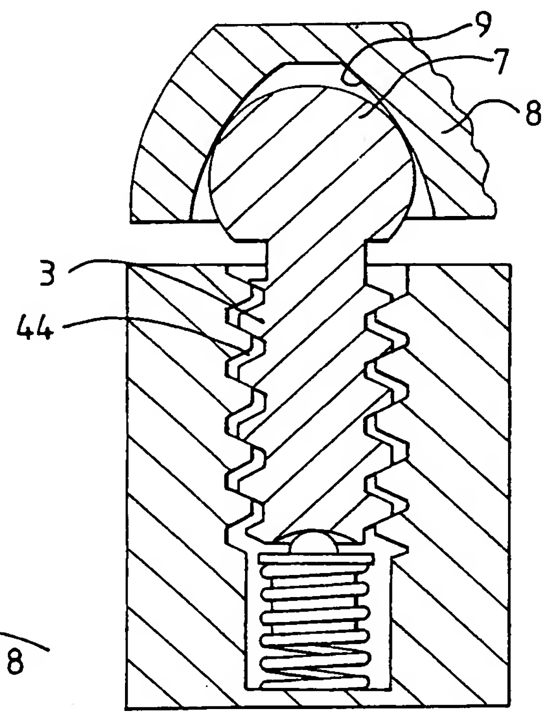


FIG 7

FIG 8

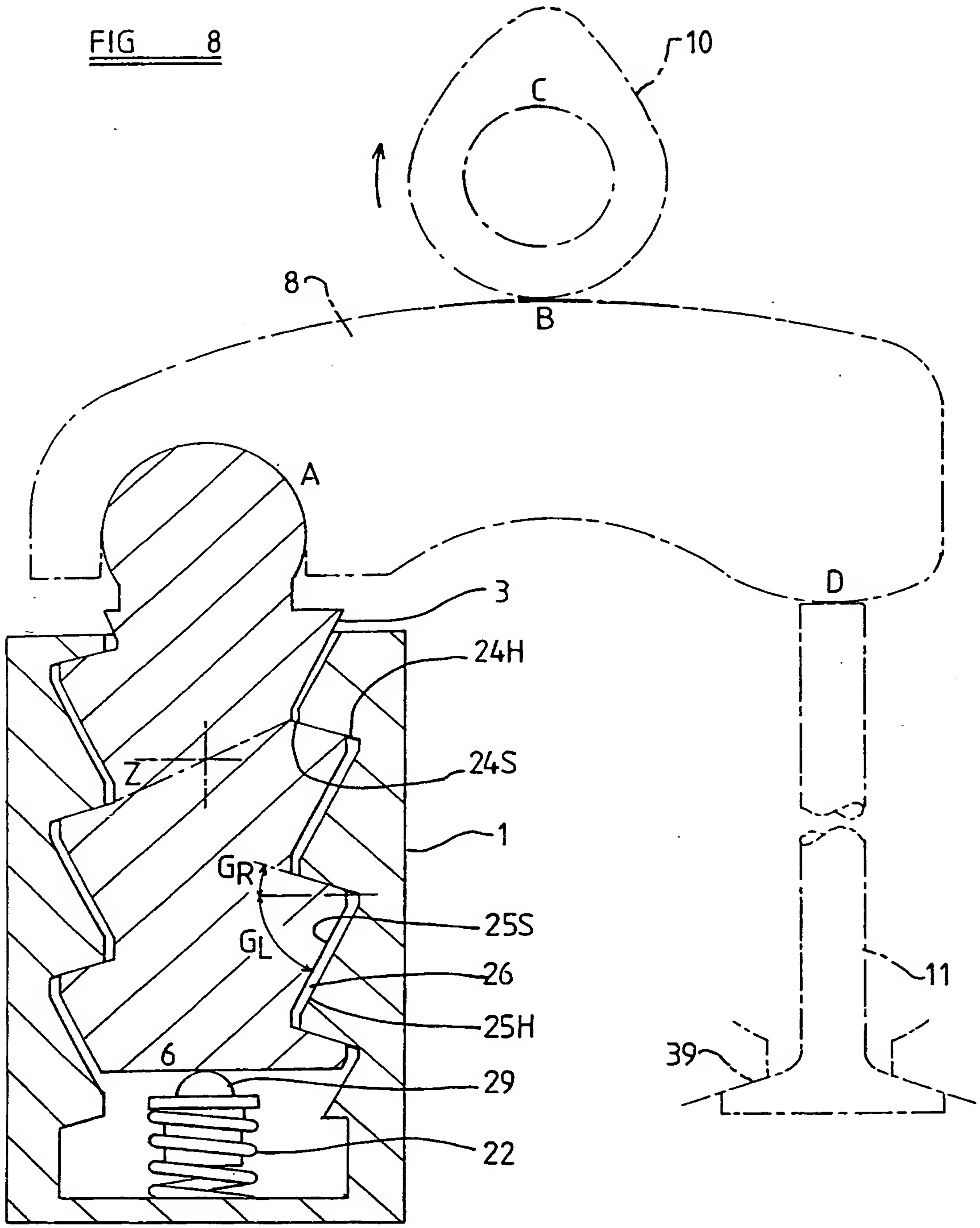


FIG 9

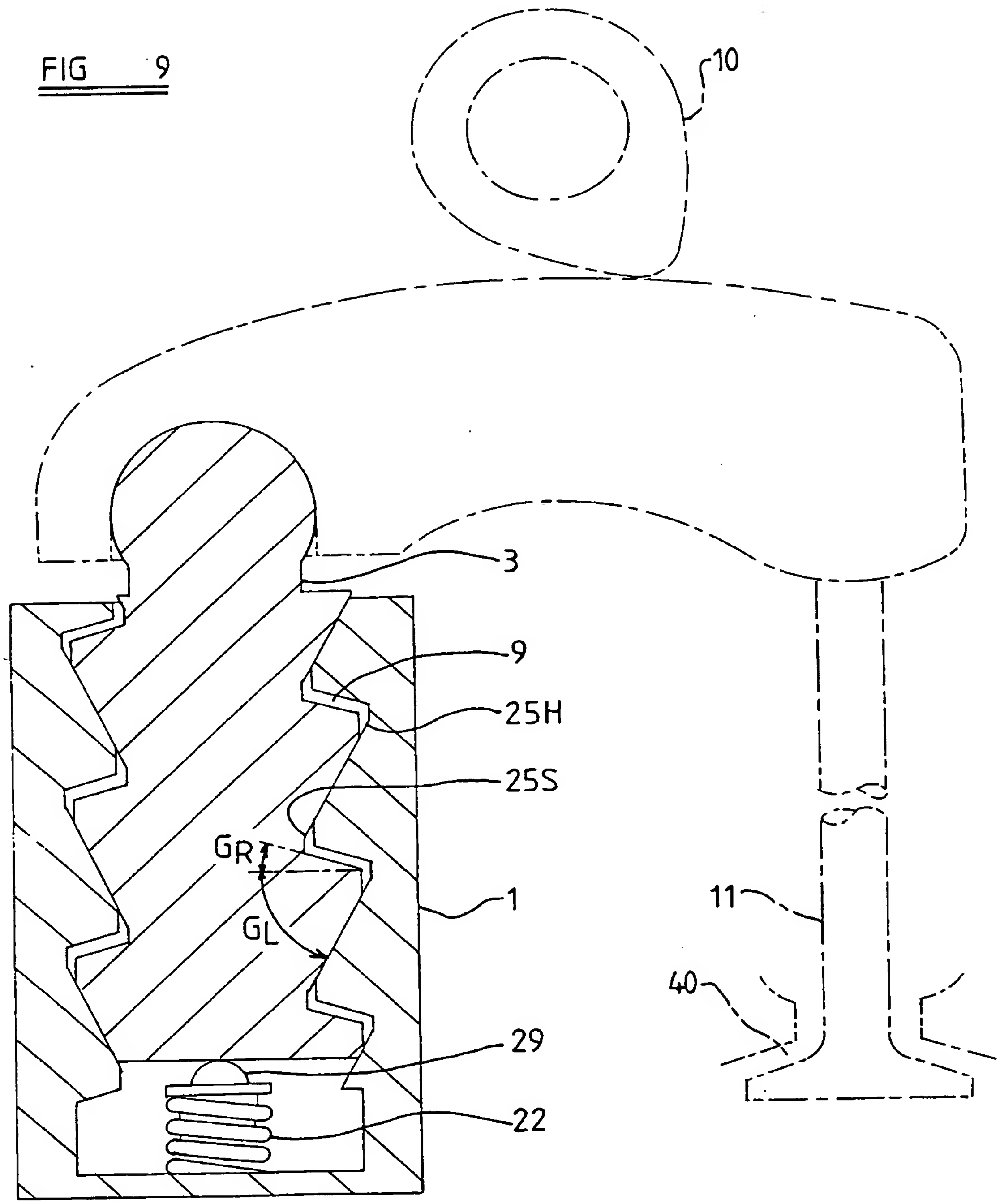


FIG 10

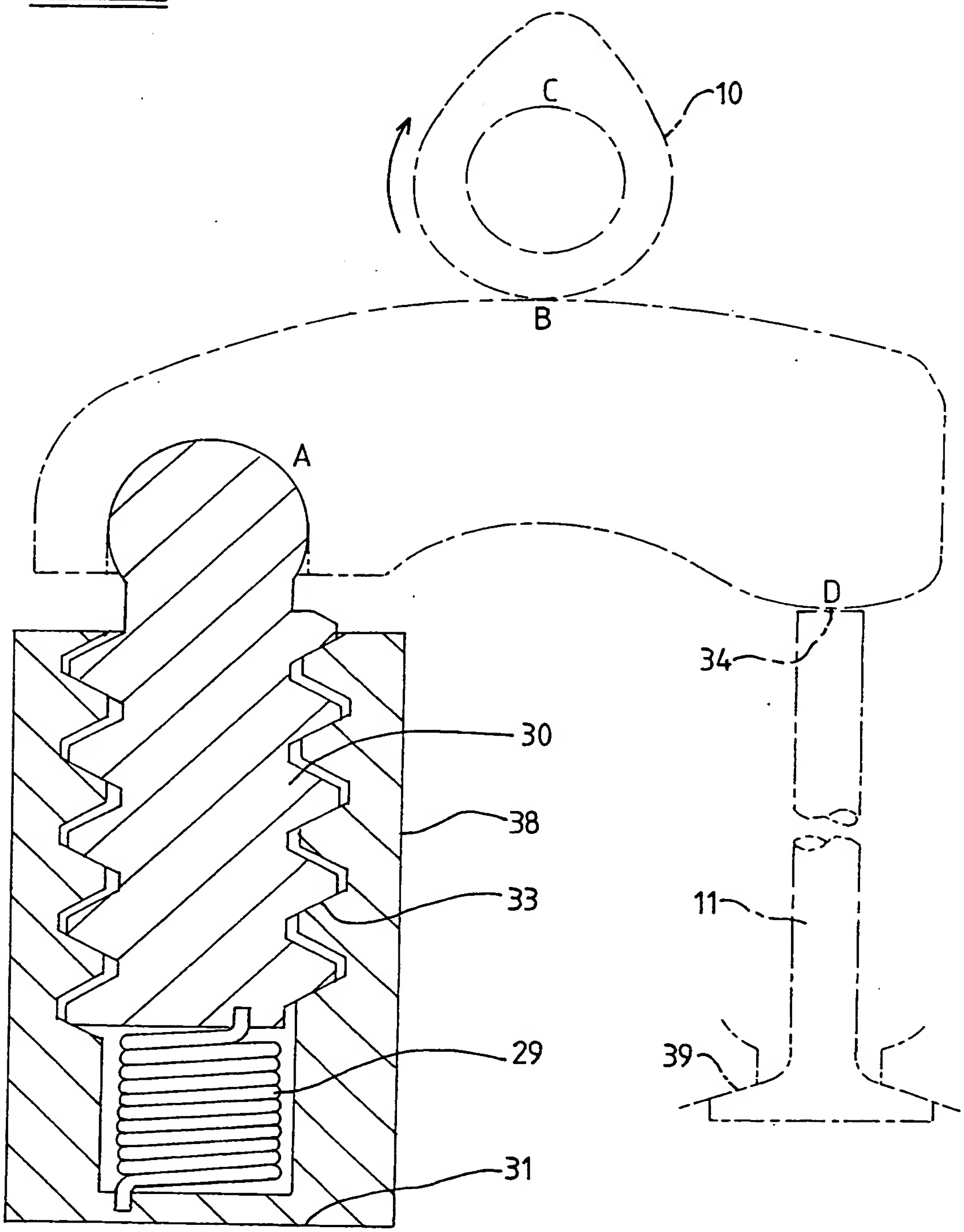


FIG 13

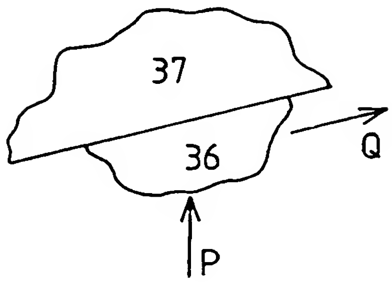
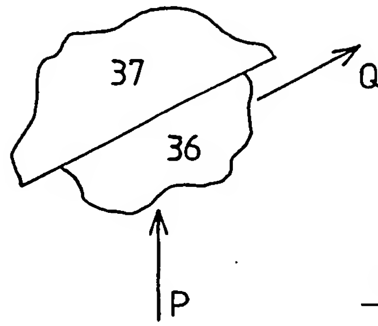


FIG 14



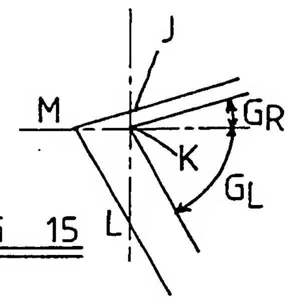


FIG 15

FIG 11

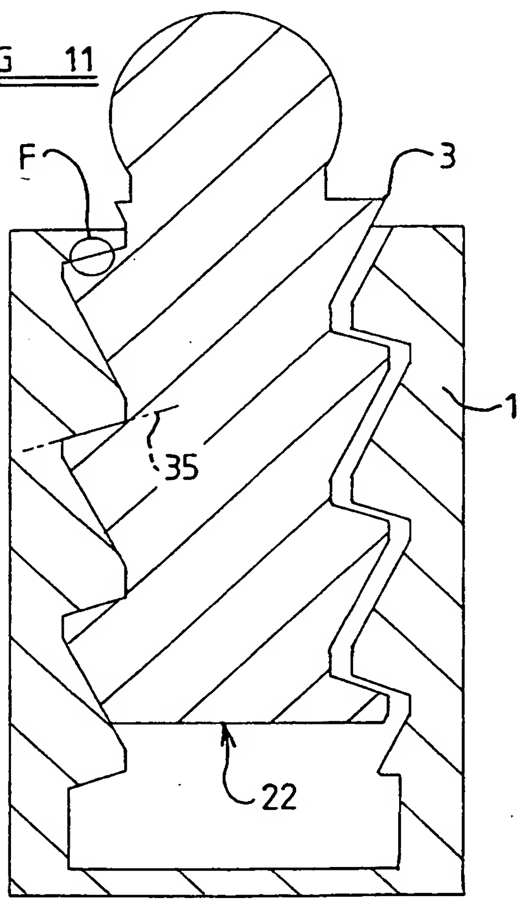
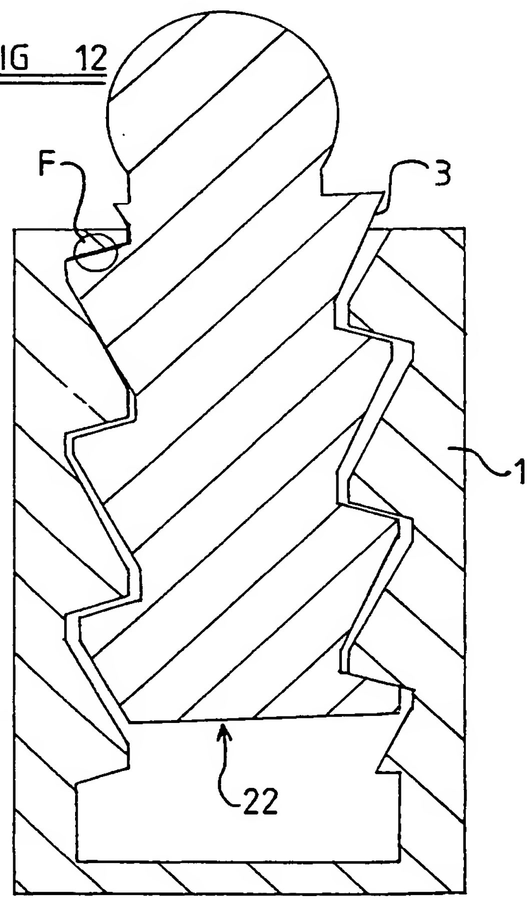


FIG 12



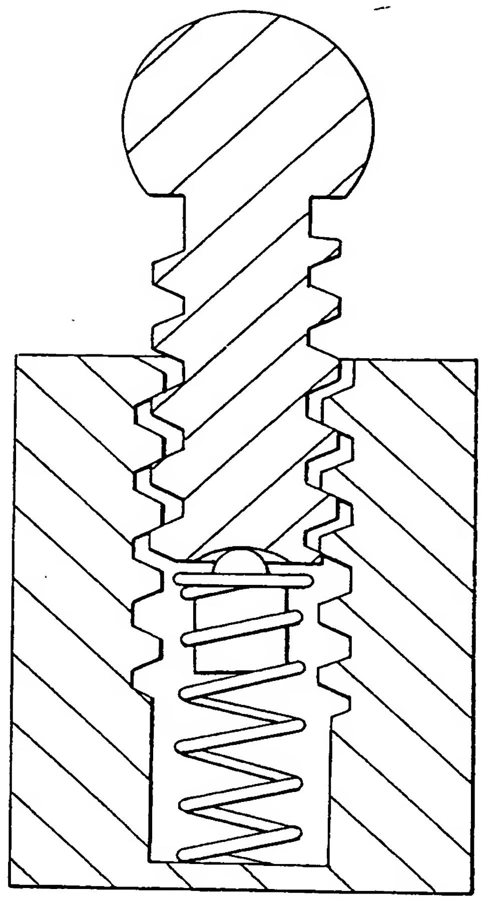


FIG 16

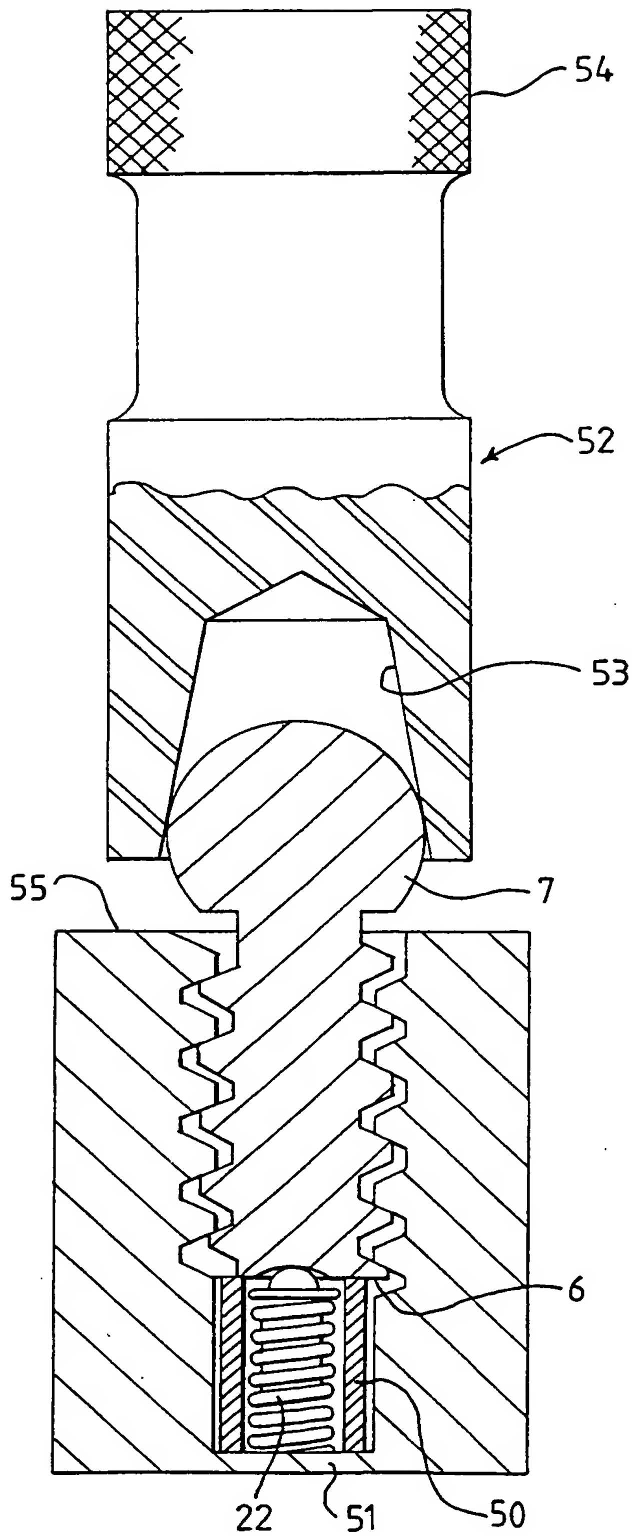


FIG 17

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(74) Agent: FORRESTER KETLEY & CO; Chamberlain House, Paradise Place, Birmingham B3 3HP (GB).

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Published:
— *with international search report*

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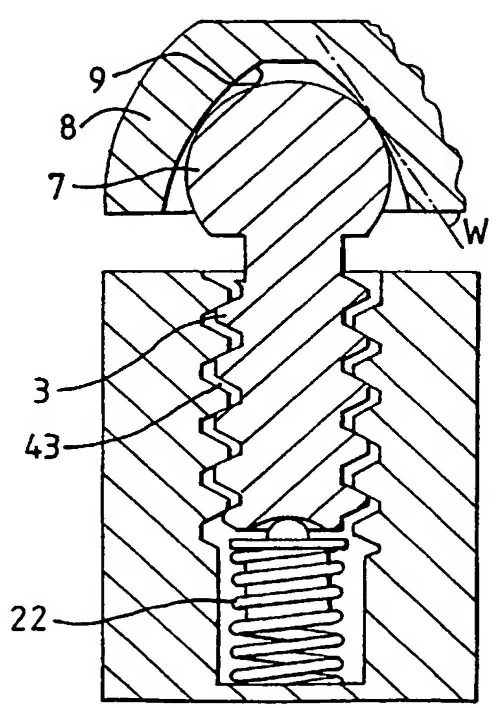
14 March 2002

For two-letter codes and other abbreviations, refer to the "Guidance Notes on Codes and Abbreviations" appearing at the beginning of each regular issue of the PCT Gazette.

(54) Title: AUTOMATIC VALVE CLEARANCE ADJUSTER



(57) Abstract: An automatic valve clearance adjuster comprising an internally screw-threaded housing (1); an externally screw-threaded screw member (3) extending within the housing; the screw thread (4) of the screw member (3) having an external form which is complementary to the thread form of the internal thread (2) of the housing and fits therein with a predetermined axial clearance, the thread being trapezoidal in form, symmetrical in axial cross-section and exhibiting equal frictional resistance against movement in both axial directions; wherein the flank angles, helix angle and number of starts of the screw thread are selected to ensure that the screw member will rotate and advance axially out of the housing solely under the influence of an axial force on its end within the housing; and the emerging end (7) of the screw member is adapted to work in conjunction with an adjacent component to receive therefrom a frictional resistance to rotation.

WO 01/57367 A3

INTERNATIONAL SEARCH REPORT

International Application No
PCT/GB 01/00410

A. CLASSIFICATION OF SUBJECT MATTER
IPC 7 F01L1/22 F01L1/24

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 7 F01L

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

EPO-Internal, WPI Data, PAJ

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	GB 1 552 786 A (GIRLING LTD) 19 September 1979 (1979-09-19) the whole document	1,10
A	WO 89 05898 A (GKN TECHNOLOGY LTD) 29 June 1989 (1989-06-29) cited in the application the whole document	1,10
A,P	US 6 032 630 A (NOZAKI TAKASHI ET AL) 7 March 2000 (2000-03-07) the whole document	1,10

☐ Further documents are listed in the continuation of box C.

☒ Patent family members are listed in annex.

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Date of the actual completion of the international search	Date of mailing of the international search report
24 July 2001	30/07/2001

Name and mailing address of the ISA	Authorized officer
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INTERNATIONAL SEARCH REPORT

Information on patent family members

International Application No
PCT/GB 01/00410

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